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ABSTRACT

This study presents effects of the vane spring stiffness on frictional power characteristics of a refrigeration compressor. Comparisons on the compressor performance are investigated for different vane spring stiffnesses with compressors designed with high and low shell pressures. Results show that while compressors with a low shell pressure using a higher spring stiffness combination may reduce frictional losses, it may also increase the starting torque requirement of the machine.

NOMENCLATURE

e	eccentricity, m
F_x	pressure differential force across vane in x direction, N
F_y	pressure differential force across vane in y direction, N
F_s	vane spring force, N
F_a	inertia force of vane, N
F_{vt}	tangential force at vane tip, N
F_{vn}	normal force at vane tip, N
F_{t1}, F_{t2}	tangential forces at vane side contact points, N
F_{n1}, F_{n2}	normal forces at vane side contact points, N
K	vane spring stiffness N/m
l	cylinder length, m
L_v	friction loss at vane tip, W
L_s	friction loss at vane side, W
L_{ec}	friction loss between eccentric and cylinder head face, W
L_{rc}	friction loss between roller and cylinder head face, W
L_{re}	friction loss between the roller and eccentric, W
M_{ec}	friction moment from eccentric to cylinder head face, Nm
M_{er}	friction moment from eccentric to roller, Nm
M_{rc}	friction moment from roller to cylinder head face, Nm
P_b, P_c	pressure in suction and compression chambers, N/m^2
P_s, P_d	suction and discharge pressures, N/m^2
R_r	roller outer radius, m
R_e	radius of eccentric, m
R_s	shaft radius, m
V_t	sliding velocity at vane tip, m/s
x	vane extension, m
θ	angular position of rotor, radian

α	offset angle of rolling piston centre, °
δ_1	clearance between roller and cylinder head face, m
δ_2	radial clearance between roller and eccentric, m
η	viscosity of lubricating oil, Ns/m^2
ω	angular velocity of eccentric, rad/s
ω_r	angular velocity of rolling piston, rad/s
$\bullet$	time differential

INTRODUCTION

In the design of hermetic rolling piston refrigeration compressors, the compressor shell is sometimes exposed to the discharge pressure (high shell pressure) and in others, to suction pressure (low shell pressure). The different in compressor shell pressure results in different starting torque characteristics. This is because the high shell pressure compressor uses the shell pressure and the spring force to ensure a good contact between the vane tip and the roller. Under this condition, during the initial machine start-up, there is a low shell pressure acting on the back of the vane hence the starting torque is lower. Once the compressor has started up, the shell pressure will be built up thereafter.

By introducing the low shell pressure, if the vane-rotor-stator features remain the same, then the back of the vane will always subject to a low shell pressure. Thus in order to maintain a good contact between the vane tip and the roller piston during the machine operation, the vane-spring force that acts behind the vane has to be increased. This increase in the spring force acts as a mean to compensate for the low shell pressure force behind the vane, as compare to the high shell pressure compressors. This action results in a constant force always acts behind the vane regardless of whether the compressor is running or not. The existence of this additional spring force during the start-up, increases the initial torque requirement, which may demand a larger starting capacitor for the driving unit.

In practice, the additional spring force requirement in the low shell pressure compressor may be introduced in two ways. One is to increase the spring compression and another is to increase the spring stiffness. The former is inadvisable because it causes large oscillating forces which contribute to a possible fatigue failure. The latter is preferred. For a proper spring design, both the static and the fatigue design criteria must be employed.

The choice of the spring stiffness, which is dependent on the shell pressure, affects the compressor power consumption, leakage losses, starting torque requirement and the reliability of the machine. The right choice of the spring stiffness is thus crucial in compressor performance and reliability. In general, too stiff the vane spring may cause unnecessary high starting torque and results in high power consumption while low spring stiffness results in chattering noise and low performance due to serious internal leakage.

A good overall design should attempt to find the required spring characteristics which fulfil both the static and the fatigue design characteristics while minimising frictional losses. This paper presents the investigation into the effects of the spring stiffness on the frictional effects of compressors with high and low shell pressures. Comparisons on frictional characteristics between a high shell pressure compressor with the low shell pressure one were made.

FRICTIONAL ANALYSIS

In rolling piston compressors, during the machine operation, the roller rolls and slides against the cylinder or stator. It also rubs against stator end faces. It rotates against the eccentric. All these, together with shaft friction contribute to the mechanical loss of the machine. To determine the friction losses of the vane-rotor-stator, the vane and the roller dynamics must be considered. The information about the vane sliding velocity, roller velocity must be obtained. There are six areas where the friction loss may take place (1,2,3,4,5).

These are friction losses occurred at the following regions (see Fig. 1a and 1b):-

i. Eccentric and the inner surface of the roller,

$$L_{re} = (\omega - \omega_r)M_{er} \tag{1.1}$$

ii. Roller face and the cylinder head face,

$$L_{rc} = M_{rc}\omega_r \tag{1.2}$$

iii. Eccentric face and the cylinder head face,

$$L_{ec} = M_{ec}\omega \tag{1.3}$$

iv. Vane tip and roller,

$$L_V = V_t F_{vt} \tag{1.4}$$

Where,

$$V_t = R_r\omega_r + e\omega\cos\theta/\cos\alpha \tag{1.5}$$

v. Vane sides and vane slot,

$$L_s = (F_{t1} + F_{t2})\dot{x} \tag{1.6}$$

vi. Outer roller surface and the inner cylinder surface

The item (vi) can be ignored if it assumes that there exists no contact forces between outer roller surface and the cylinder. More detail analysis may be referred to [5].

Where the last unknown ω_r may be obtained from eqn. (1.7) if a simplified analysis assuming steady roller rotation applicable.

$$\omega_r = \frac{\left(2\pi\eta\omega l R_e^3/\delta_2 - R_r F_{vt}\right)\delta_2\delta_1}{2\pi\eta\left(lR_e^3\delta_1 + \left(R_r^4 - R_e^4\right)\delta_2\right)} \tag{1.7}$$

RESULTS AND DISCUSSION

For the purpose of comparison, the initial study was carried out to search for a suitable spring stiffness for an existing high shell compressor if it were to operate at low shell pressure. The criteria defined to fulfil the above task was to find the spring with stiffness that could provide positive contact at the vane tip and roller through out the operating cycle. Fig. 2 shows the vane tip contact force for a high shell pressure compressor with K=1101N/m compared with various spring stiffness of a low shell pressure compressor, operating under identical operating conditions. The vane tip reaction force is the most crucial one. Generally, in the design of this type of compressor, sufficient vane tip contact force is required to maintain good sealing between the low and high pressure chambers. Too high the vane tip contact force causes excessive frictional losses whereas too low the force causes vane chattering and severe leakage passes through the contact region. The negative contact force in the figure indicates the departure from the physical contact. The results , see Fig. 4, show that in order to have reasonably good vane tip contact, the spring constant for the low shell pressure model must be at least 4 times the previous value for the high shell model.

Preliminary tests conducted on a low shell pressure compressor, revealed that when using spring stiffnesses of less than the minimum requirement to produce the positive vane-tip roller contact, showed serious vane chattering and reduced capacity. Tests conducted also showed that higher spring stiffnesses cause high starting torque requirement. Fig. 3 shows the variations of the spring force alone, for high and low shell pressure compressors, for a complete roller rotation. It is suspected that, the higher spring force (even at the initial start-up) for the case of low shell pressure compressor, may have contributed to the higher starting torque requirement. Fig. 4 shows the net pressure force acting along the direction favourable to vane-tip roller contact. It shows that, in general, the pressure force in the high shell pressure compressor is always in favour of the vane-tip roller contact, whereas in the case of the low shell pressure compressor, the pressure force may against the vane-tip roller contact, especially during the discharge process, when the chamber pressure is high, this is depicted y negative pressure force in the figure.

The modelling results also show that the contact force is averaged about 30 N lower in the case of low shell pressure combination with a low spring stiffness. This lower contact force results in a lower vane tip-roller frictional loss, but it results in a higher roller-cylinder head faces friction and higher roller-eccentric friction, see Fig. 5, 6 and 7. The increase in the latter is found to cause by an increase in the angular speed of the roller due to lower vane tip roller contact force., see Fig. 8.

Fig. 9 shows the comparison of the total friction loss. In general, low shell pressure compressors show lower friction losses.

CONCLUSION

The study shows that while compressors with low shell pressure using a higher spring stiffness combination may marginally reduce frictional losses thus improving the mechanical efficiency, it may also increase the starting torque requirement of the machine.

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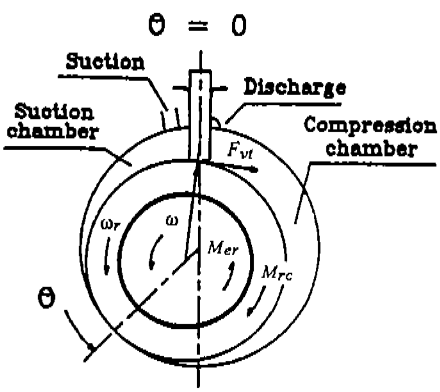

Fig. 1 a Roller firction moments

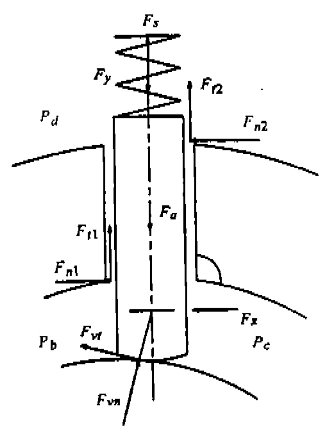

Fig. 1b Vane force balance

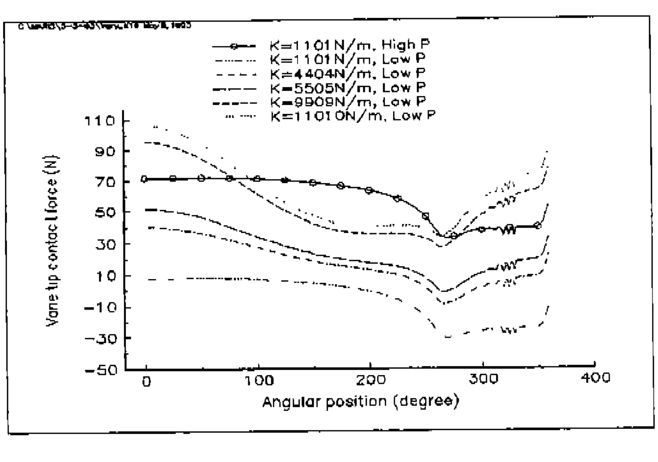

Fig. 2 Variation of vane-tip contact force

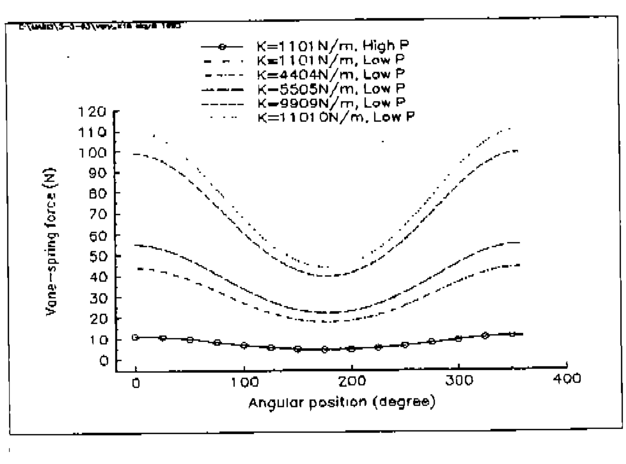

Fig.3 Vane spring force with spring stiffnesses

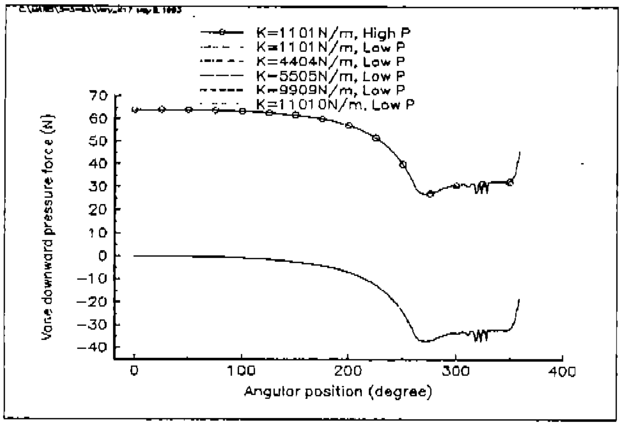

Fig. 4 Net downward pressure force on the vane

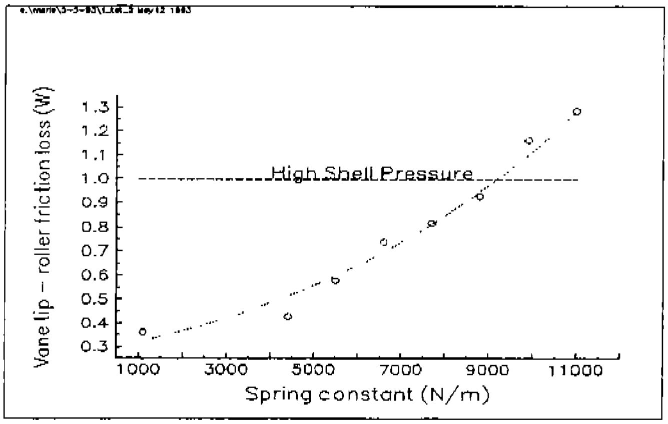

Fig. 5 Vane tip-roller friction with spring stiffnesses

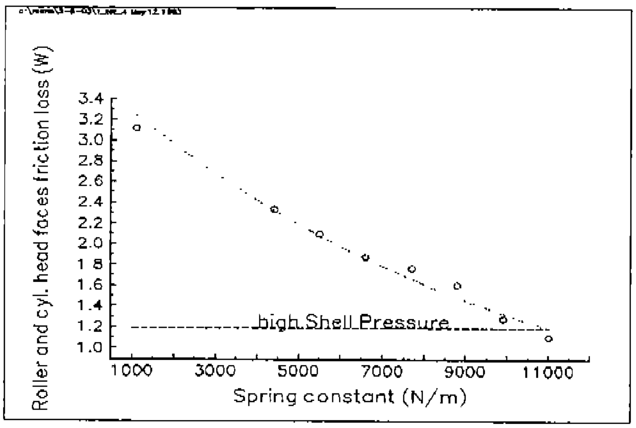

Fig. 6 Roller-Cly. head face friction with spring stiffnesses

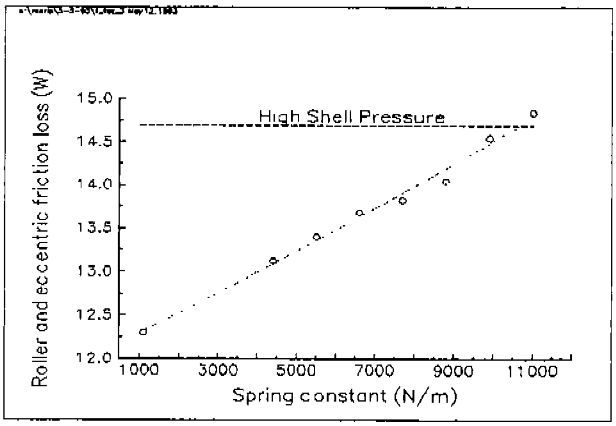

Fig. 7 Roller-eccentric friction with springstiffnesses

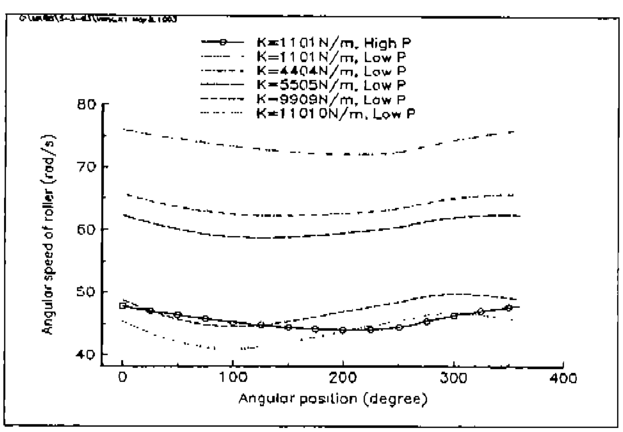

Fig. 8 Angular speed of the roller

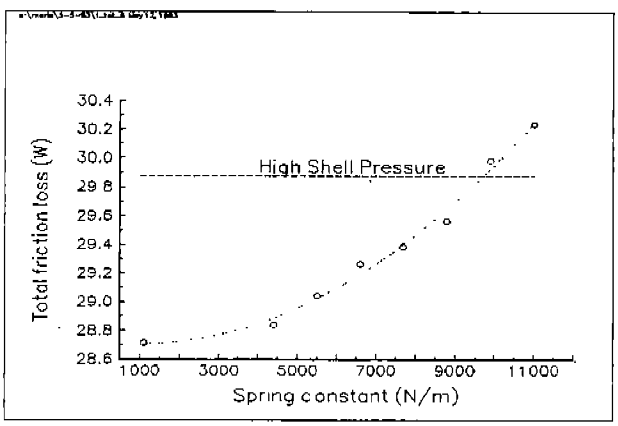

Fig. 9 Total friction loss with springstiffnesses